



# Heat treatment Elimination in Forged steel Crankshaft of Two-stage Compressor

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## Abstract

*There are various components in reciprocating compressor; however, optimization of compressor crankshaft for design and manufacturing is planned by considering its volume and cost. The crankshaft used in this study is forged steel from a two-stage air compressor models. This is a single throw crankshaft consists of one drive end shaft (fly end), one non-drive end shaft (free end) and a crank pin. High strength, ductility, and fatigue resistance are critical properties required from the crankshaft material and manufacturing process. In spite of its generally lower fatigue resistance, ductile cast iron is a major competition to forged steel crankshafts in the competitor's compressors. Moreover, there is no heat treatment process involved in case of ductile cast iron. Since a crankshaft experiences a large number of load cycles during its service life, fatigue performance and durability of this component is a key consideration in its design and performance evaluation. The design and manufacturing including heat treatment process of forged steel crankshaft is studied for optimization and to reduce cost. Experimental investigations made to evaluate the mechanical properties of the crankshaft material with and without heat treatment and compared. Design verification made through static and dynamic analysis using finite element software to find the factor of safety for both cases. Crankshafts are manufactured and it is assembled in the compressors and validation is done through accelerated tested at in-house as well as in the field to prove the process. The cost benefit analysis is done for manufacturing of the crankshaft without heat treatment.*

**Keywords:** Crankshaft, compressor, heat treatment.

## Introduction

The increased demand for improved performance and reduced cost in compressors has lead to significant competition in compressor components materials and manufacturing process technologies. In addition, the stiff cost competition in the market due to many new players and steep increase of commodity price in the market have created a need to look at the cost to maintain the profit levels.

In reciprocating compressor transmission system consists of crankshaft, main bearings, connecting rod and coupling connected to the prime mover. In this design, the compressor crankshaft drives the compressor as well as it is used to transmit the power to other accessories like traction motor blower and radiator fan in diesel locomotives. Crankshaft is a large component with a complex geometry in the compressor transmission system, which converts the rotary motion of the shaft to reciprocating displacement of the piston with a four-link mechanism. This analysis was conducted on a two-cylinder two-stage compressor.

Crankshaft experiences large forces from first and second stage cylinders during loading and unloading of the compressor<sup>1</sup>. The power is transmitted to the crankshaft through pulley or coupling connected from electric motor. As the crank pin is

rotated, the connecting rod converts the rotary motion to reciprocate the piston. The magnitude of the force depends on many factors that consist of crank radius, connecting rod dimensions, weight of the connecting rod, piston, piston rings, and pin. Gas forces in the top of the piston and inertia forces acting on the crankshaft cause two types of loading on the crankshaft structure; torsional load and bending load.

Due to the crankshaft geometry and compressor mechanism, the crankshaft fillet experiences a large stress range during its service life. Figure 1. Shows a crankshaft in the engine block from side view. In this figure, it can be seen that at the moment of combustion the load from the piston is transmitted to the crankpin, causing a large bending moment on the entire geometry of the crankshaft. At the root of the fillet areas stress concentrations exist and these high stress range locations are the points where cyclic loads could cause fatigue crack initiation and leading to fracture.

The crankshaft fillet rolling process is one of the commonly adopted methods in engineering to improve fatigue life of the crankshaft<sup>2</sup>. Compressive residual stresses on and below the fillet radius surface are induced through the fillet rolling operation. Consequently, fatigue life of the crankshaft is improved. This is quantified experimentally and analytically. A non-linear finite element analysis was implemented to simulate

the crank shaft rolling process. A crack modeling technique was developed to calculate the equivalent stress intensity factor range based on the combined residual stresses and operating stresses along each potential crack growth plan.



**Figure-1**  
**Crankshaft configuration**

The fatigue limit of rolled, ductile cast iron crankshaft sections under bending was investigated experimentally and analytically<sup>3</sup>. The relationship of different failure criteria – specifically, surface crack initiation, resonant shifts, and two-piece failures – on the fatigue limit of the crankshafts was explored from testing. A comparison showed that using the surface crack failure criterion decreased the apparent fatigue limit of the crankshaft significantly. This result supported the theory of crack arrest in the fillet area of crankshafts and raises a significantly.

The geometry of the crankshaft was geometrically restricted due to limitations in the computer resources available to the authors<sup>4</sup>. The FEM analysis was performed in ANSYS software and a three dimensional model made of Photo elastic material with the same boundary conditions was used to verify the results. This study was based on static load analysis and investigated loading at a specific crank angle. The FE model results showed uniform stress distribution over the crank, and the only region with high stress concentration was the fillet between the crank-pin bearing and the crank web.

**Design verification:** The design calculation is made by using heat-treated crankshaft mechanical properties and without heat-treated crankshaft mechanical properties. The below conditions are applied to calculate the diameter of the crankshaft, i. Shaft subjected to twisting moment only, ii. Shaft subjected to bending moment only, iii. Shaft subjected to both bending and twisting moment.

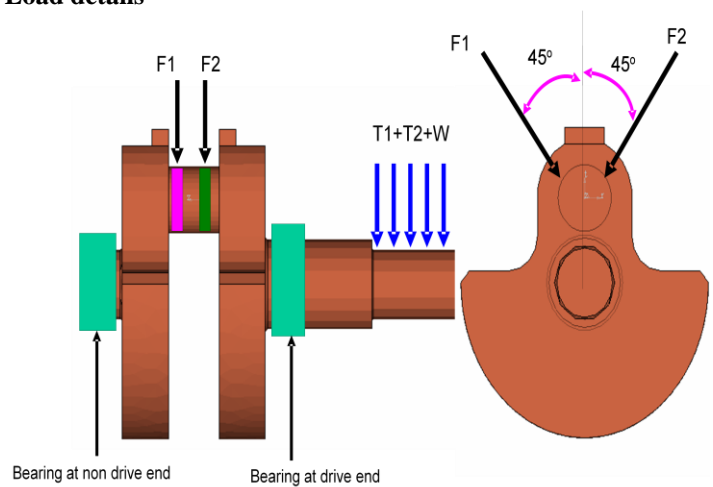
In all the above three conditions the diameter is calculated and compared with actual dimension and found same. The factor of safety of the design is compared for both and found as below: i. Factor of safety of crankshaft design with heat treatment = 7.6, ii. Factor of safety of crankshaft design without heat treatment = 5.5.

The crankshaft design is safe even without heat treatment and enhanced mechanical properties. The Load details are calculated<sup>5</sup> by considering the gas forces, pulley mass and belt tension as this compressor crankshaft being used in belt driven compressors as shown in figure 2.

The crankshaft investigated in this study is shown in figure 1 and belongs to compressor with the configuration shown in table 1. The resultant load acting in the crankpin from each cylinder calculated with respect to crankshaft angle shown in figure 3. Although the pressure plot changes for different compressor speeds, the maximum pressure, which is much of our concern, does not change and the same graph could be used for different speeds. The geometries of the crankshaft, bearing assembly and connecting rod from CAD software, which provided the solid properties of the connecting rod such as moment of inertia and center of gravity (CG). These data were used in to calculate inertial loads and total load on each connecting rod big end bearings against the crankshaft angle.

Finite element analysis was performed to obtain the stress magnitude at critical locations at worst load cases. The pressure-volume diagram was used to calculate the load boundary condition in dynamic simulation model, and other simulation inputs were taken from the compressor specification. The dynamic load calculation is done analytically by considering the reciprocating masses and its inertial loads in all cylinders. This load was applied to the FE (Finite element) model in ANSYS software, and boundary conditions were applied according to all loads acting on the crankshaft mounting. The stresses are calculated from FE model and plotted in soderberg diagram to calculate the factor of safety. Results achieved from aforementioned analysis can be used in fatigue life calculation and optimization of this component.

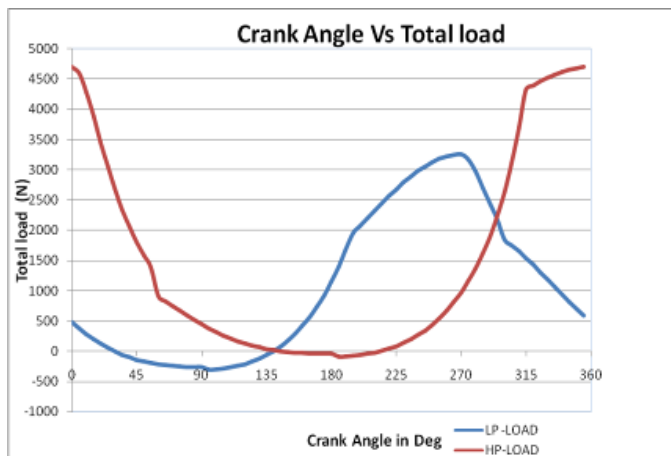
**Load details**



**Figure-2**  
**Cylinder orientation and its location**

**Table-1**  
**Configuration of the compressor**

Parameter	First stage	Second stage	units
Crank radius	4.2	4.2	cm
Diameter	10.0	6.0	cm
Max pressure	3.5	12.0	bar
Connecting rod length	18.0	18.0	cm
Distance of the CG of the connecting rod from crank end canter	4.25	4.25	cm
Mass moment of inertia of connecting rod	5.2 x 10 <sup>4</sup>	5.2 x 10 <sup>4</sup>	Kg mm <sup>2</sup>
Mass of the connecting rod	2.65	2.65	kg
Mass of piston assembly	1.210	0.556	kg



**Figure-3**  
**Total load versus crankshaft angle**

**Finite Element modelling and analysis:** The finite element model of the crankshaft was developed using ANSYS. Analysis was carried using Non linear quasi static approach and crankshaft geometry is meshed with second order tetrahedral elements. As a crankshaft is designed for very long life, stresses must be in the linear elastic range of the material. Therefore, all carried analysis are based on the linear properties of the crankshaft material. The meshed crankshaft with tetrahedral elements is shown in figure 4.

Crankshaft was analyzed as a part placed in between the bearing and the operational loads imposed as tangential and normal components. Gas pressure loads, fan torque and the inertial loads of the reciprocating piston assembly was considered in the simulation. The mesh of the finite element model used to evaluate the stiffness of the crankshaft is shown in figure 4. The same model was also used in the final step to evaluate the stress level in the crankshaft. The boundary conditions applied during the evaluation of the stiffness were null radial displacement in bearing regions.

**Boundary conditions:** Boundary conditions in the FE model were based on the compressor configuration. The mounting of this specific crankshaft is on two different bearings which results in different constraints in the boundary conditions. Two rigid rings used to simulate the bearing constraint in all degrees of freedom. This indicates that the surface cannot move in either direction and cannot rotate. Structural coupling elements used to constraint fly end in rotational degrees of freedom and fan load applied at the non-drive end as torque. It should be noted that the analysis is based on dynamic loading, though each finite element analysis step is done in static equilibrium. The main advantage of this kind of analysis is more accurate estimation of the maximum and minimum loads. Design and analyzes of the crankshaft based on static loading can lead to very conservative results. In addition, as was shown in this section, the minimum load could be achieved only if the analysis of loading is carried out during the entire cycle. The stress distribution in the crankshaft is shown in the figure 5 for one load case.

As the dynamic loading condition is analyzed, six load steps are applied on the surface of the crankpin bearing due to gas forces in the low pressure and high-pressure cylinders along with other loads like fan torque and inertial loads.

For ductile materials, failure Analysis is done based on the Soderberg line<sup>6</sup> as shown in figure 6 which is the line between Yield Strength and Endurance Strength of the material. The Line of safe stress is drawn based on fatigue factor of safety considered. Mean and Amplitude Stresses is plotted at various nodes of critical locations of the shaft.

## Methodology

In experimental work, two specimens, one with heat treatment and one without heat treatment is prepared as per IS 210 and tensile testing carried out to compare the tensile strength, shear strength and percentage of yield as shown in the table 3 The compressor is assembled with, non heat-treated crankshaft and accelerated test is carried at in house for 2000 hours. The compressor is opened in every 500 hours of interval, checked for run out in oil seal location, connecting rod parallelism and distance between main bearings, and found within specifications.

## Results and Discussion

From the FE static and dynamic analysis the design is verified for the crankshaft, which is manufactured without heat treatment for the two stage oil flooded compressor up to 25 HP capacities. Some locations on the geometry were considered for depicting the stress history. These locations were selected according to the results of FE analysis and all the selected elements are located on different parts of the fillet areas due to the high stress concentrations at these locations. Selected locations are labeled, maximum and minimum principle stress are extracted from the from the stress plot .The mean and

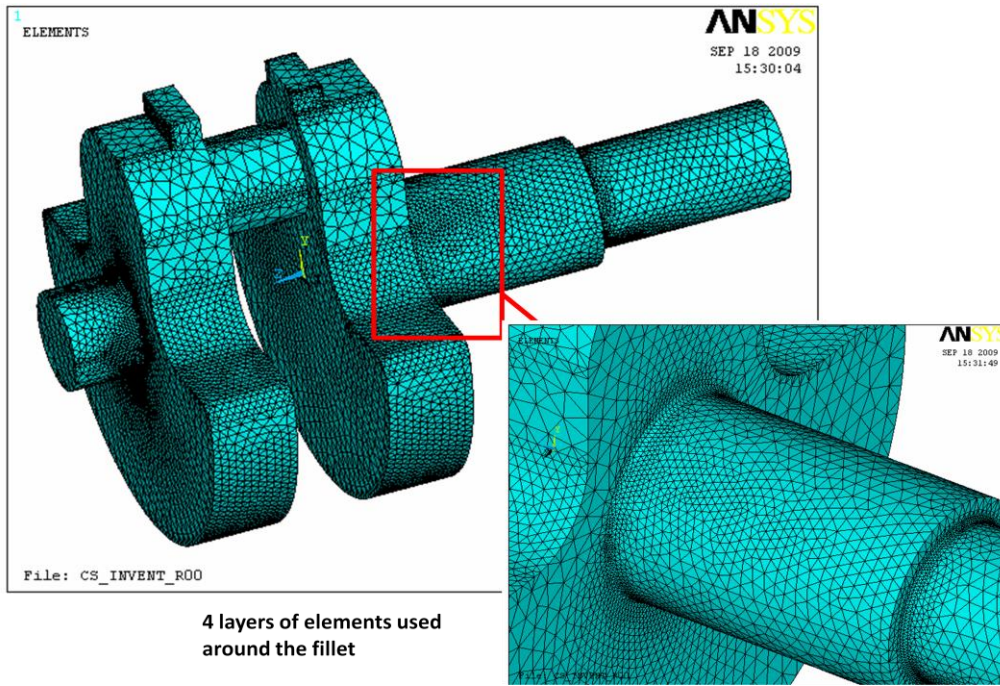
amplitude stresses are calculated to plot the soderberg diagram and endurance strength as shown in figure 6.

From the field validation, the crankshafts are disassembled and fit was checked between the main bearing locations by pulling the bearing from its position. The distance between the main bearing is measured and found within specification. The connecting rod play and run out in the oil seal seating area are

measured and found within the specification. The field validation is continued for more than one year to prove this crankshaft in the two-stage compressor in different applications.

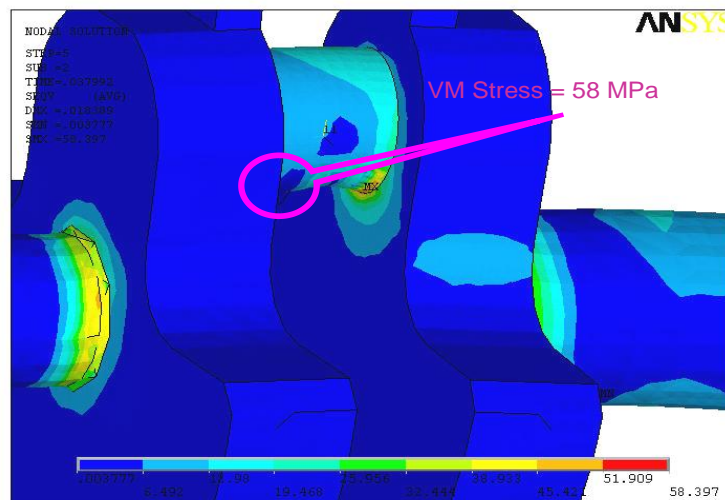
Field validation is carried out in different locations of the country and different applications as shown in table 4.

**Finite Element model**



4 layers of elements used around the fillet

**Figure-4**  
**FE model with tetrahedral elements**



**Figure-5**  
**Boundary conditions applied in FEA model**



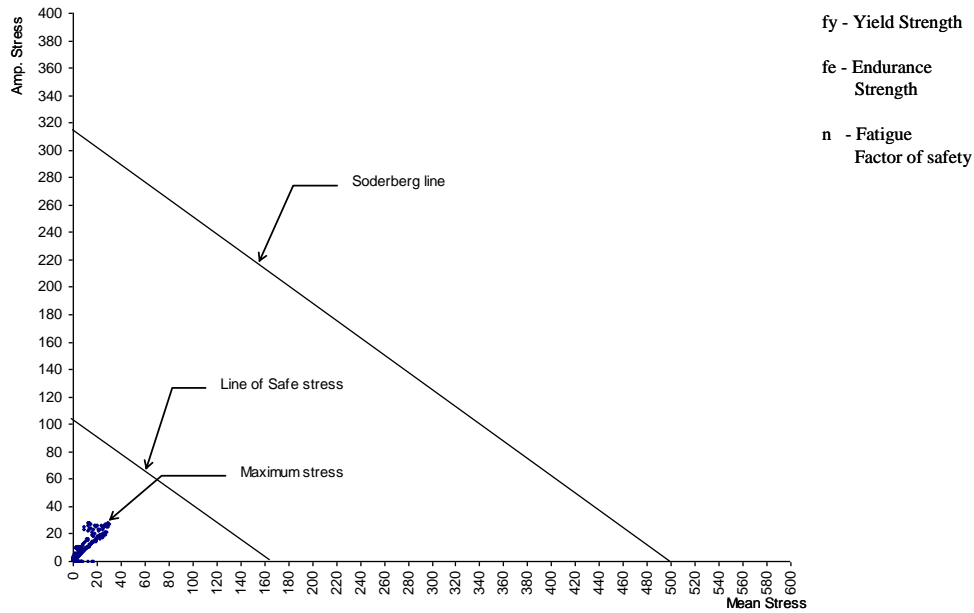


Figure-6

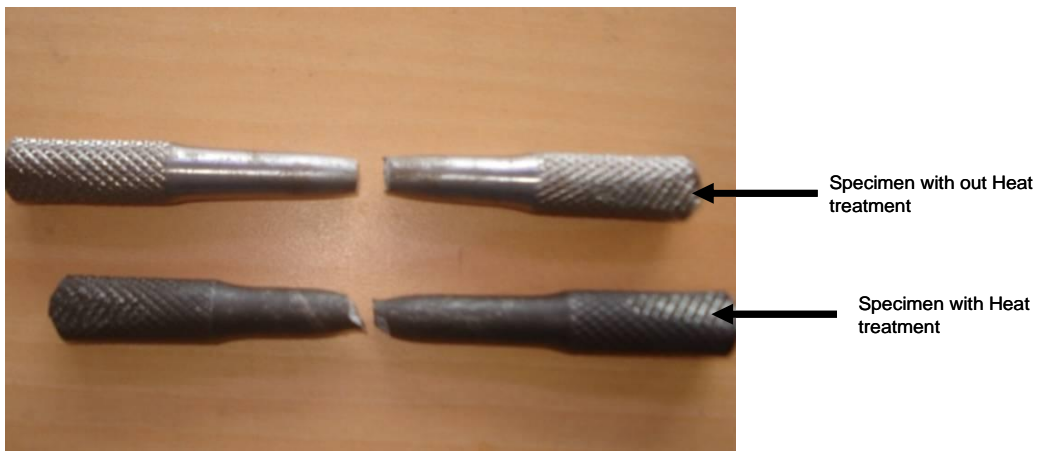


Figure-7  
 Soderberg diagram

Table-2  
 Results from FE analysis

Parameter	Maximum value obtained from FEA	Yield Strength	Location
Von Mises Stress	58 Mpa	499 Mpa	HP connecting rod side of the Crank pin
Total Deflection	0.025 mm	-	Pully mounting area of the Fly end shaft

Table-3

Parameter	Measured Value	
	With heat treatment	Without heat treatment
Tensile strength (MPa)	995	633
Yield strength (MPa)	699	499
Shear strength (MPa)	350	250
Percentage of Elongation	14.6	21.6

Table-4

FIELD VALIDATION DETAILS					
Application	Working atmosphere	Location	No of units	Hours of validation	Duty cycle
1.Cement industry 2.Plastic industry 3.Mining industry	1.High ambient 2.Dusty areas	Delhi	10	3000 to 3500 hours	70 % on and 30% off
		Rajasthan	5		
		Chennai	3		
		Ahmedabad	3		
		Cochin	3		
		Coimbatore	5		
		Kolkata	2		
Hyderabad	3				

**Conclusion**

Theoretical, experimental and field validation results are favour of using the forged crankshaft in the two stage oil flooded compressors. Therefore, crankshaft is manufactured by eliminating the heat treatment process in the production line to reduce the work in progress (WIP) inventory, lead-time, space reduction and single piece production flow from machining, grinding, assembly and testing. The cost benefit analysis is carried out by considering the process time and inventory and it is estimated about 29.5 lakhs per year by considering the annual production. Therefore, we can conclude that, this forged crankshaft material can used without hardening and tempering for the two-stage oil flooded compressors up to 25 HP capacity.

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